RATIONAL DESIGN OF FAST BOAT HULL

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Abstract. We present a new method for the rational design of the hull structure of planing boats, which considers hydro-elasticity, structural dynamics and nonlinearity of geometry and material. The method combines rules calculations with analytical expression and analysis of FSI to a practical design procedure. A design example demonstrates a saving of 20% of the bottom plate thickness, relative to design by rules, while keeping the rules allowable stress. Exceeding the rules allowable stresses enables further reduction of weight. We designed and constructed a full scale research boat and held sea trials, measuring strains at high sampling rate. Our research boat has two sides of different construction: the port side is designed by rules, while the starboard side is designed by our rational design method, with 20% thinner plates and double spacing between the longitudinal stiffeners. We present a verification of our design method: A comparison of stresses between design by rules, rational design, and measurements in the sea trials shows: For the heavy side (designed by rules), rules, rational, and trials show similar stresses, so both rules and rational are applicable for design; While for the light side (rational design), the rules dramatically over assess the stresses, while rational and trials show good agreement. We therefore expect this study to advance the design practice, to obtain more efficient boats.

1 INTRODUCTION

Typically, the dominant load for the design of planing hulls is slamming, while sailing fast at head seas. Classification rules, such as [1,2,3,4], assess the structure by statically applying an empirical design pressure to the structural members, represented by linear beam theory. However, the slamming is a violent Fluid Structure Interaction (FSI), where hydro-elasticity and dynamic structural responses are important [5]. For a rigid boat, the slamming pressure may be assessed by analytical solution, developed by Von Kerman [6] and extended by Wagner [7]. To consider hydro-elasticity, as well as nonlinear structure (geometry and material), the water entry problem is being solved by numerical methods, such as: Arbitrary Lagrangian–Eulerian (ALE) Finite Elements (FE) formulation, see for example [8], and Smoothed Particle Hydrodynamic (SPH), see for example [9]. Although hydro-elastic water entry is widely studied by researchers, it is yet rarely considered by boat designers. The design by rules of offshore fast boats typically results with robust hull. This research offers a design method [10] for the hull structure of planing vessels, which considers hydro-elasticity and nonlinear structural dynamics. Our method combines rules, theoretical solutions and numerical analysis to a practical design procedure and leads to more efficient design.

Section 2 presents the design procedure. Section 3 presents the design and construction of our research boat. Section 4 presents preliminary results of the sea trials and a comparison between design by rules, rational design by our method and measurements at the sea trials. Section 5 presents our conclusions and direction of continuance study.

2 RATIONAL DESIGN PROCEDURE

The design procedure is presented in [10]. We follow it here with an example, which is the design of our research boat.

2.1 Preliminary design of the hull by applicable classification rules

For the design of our research boat we adopted RINA rules [4]. Here we indicate the design parameters used for the Port side, which is designed by rules, while at the Starboard side the scantlings is reduced based on our rational design method. Our method considers structural design, while hull design includes other aspects as: functional architecture, hydrostatics, and hydrodynamics. Hence, we focus on the structural design; while only mention other aspects for the sake of completeness. The relevant steps of design by rules are:

(a) Specify a design speed, at a related sea-state: *24knots* at significant wave height *1m*.

(b) Preliminary design the hull and specify the geometrical parameters required to assess scantlings by rules: waterline length *7.2m*, greatest molded breadth *2.5m*, deadrise-angle (measured from the water plane to the V bottom) at the center of gravity *20.3°*.

(c) Assess weights and hydrostatics and specify mass parameters, required for the structural assessment by the rules: displacement *3000kg*, LCG (Longitudinal Center of Gravity) *2.8m*, draft *0.45m* and running trim *4°*.

(d) Apply rules to assess quasi static design pressure. The calculations by rules assess maximum stresses in the structural members (plates, longitudinal stiffeners, transverse frames) and determine the scantlings accordingly. The rules apply beam bending theory, where the critical load is typically the slamming pressure, which is uniformly distributed along each "beam" and is statically applied. Each "beam" may be a strip of the bottom plate between two stiffeners, or a longitudinal stiffener between two transverse frames or a transvers frame between the hull sides. The slamming pressure is proportional to the design vertical acceleration at the *LCG*:

$$a_{CG} = \frac{(50 - \beta_{CG})\left(\frac{\tau}{16} + 0.75\right)}{3555 C_B}\left(\frac{H_S}{T} + 0.084\frac{B_w}{T}\right) K_{FR} K_{HS}, \tag{2–1}$$

where: β_{CG} is the deadrise angle in degrees at *LCG* and should be taken at the range of *10°* and *30°*; τ is the running trim angle in degrees and should be taken at least *4°*; H_S is the significant wave height in meters; *T* is the fully loaded draft in meters, at rest in calm water; B_w is the greatest moulded breadth measured on the waterline at draft *T*; $C_B = \frac{\Delta}{\rho L B_w T}$ is the block coefficient, related to the waterline length *L* in fully loaded displacement Δ, the density of the water ρ in ton/m^3, 1.025 for sea water; K_{FR} and K_{HS} are coefficients defined by:

$$\begin{cases} K_{FR} = \left(\dfrac{V_x}{\sqrt{L}}\right)^2 \; ; \; K_{HS} = 1 \quad , \quad if \;\; \dfrac{\Delta}{(0.01L)^3} \geq 3500 \;\; and \;\; \dfrac{V}{\sqrt{L}} \geq 3 \\ K_{FR} = 0.8 + 1.6\left(\dfrac{V_x}{\sqrt{L}}\right) \; ; \; K_{HS} = \dfrac{H_s}{T} \quad , \quad if \;\; \dfrac{\Delta}{(0.01L)^3} < 3500 \;\; or \;\; \dfrac{V}{\sqrt{L}} < 3 \end{cases} \tag{2–2}$$

In (2–2) V is the maximum service speed and V_x is the actual boat speed, in knots.

Table 2-1 summarizes the parameters of our research boat, for which $a_{CG} = 4.97g$.

$\boldsymbol{V_x}$	H_S	$\boldsymbol{\tau}$	$\boldsymbol{\beta_{CG}}$	$\boldsymbol{B_W}$	$\boldsymbol{T}$	$\boldsymbol{L}$	$\boldsymbol{\Delta}$	$\boldsymbol{C_B}$	$\boldsymbol{\rho}$
24Knot	1.0m	4^0	20.3^0	*2.5m*	*0.45m*	*7.2 m*	*3 ton*	*0.36*	$1.025\ ton/m^3$

Table 2-1: Boat Design parameters

The maximum slamming pressure on the bottom of the hull is given by (2–3) in *kPa*:

$$p_{sl} = 70\frac{\Delta}{S_r}K_1K_2K_3a_{CG}, \qquad S_r = 0.7\frac{\Delta}{T} \tag{2–3}$$

In (2–3): S_r is a reference area, in m^2, K_1 is a factor for longitudinal distribution, defined by equation (2–4), where x is the distance, in m, from the aft perpendicular to the load point.

$$\begin{cases} K_1 = 0.5 + \frac{x}{L} \; , & \frac{x}{L} < 0.5 \\ K_1 = 1 \; , & 0.5 \leq \frac{x}{L} \leq 0.8 \, , \\ K_1 = 3 - 2.5\frac{x}{L} \; , & 0.8 < \frac{x}{L} \end{cases} \tag{2–4}$$

K_2 is a factor related for the impact area, defined by (2-5) and must be grater then: *0.5* for plating, *0.45* for stiffeners and *0.35* for girders and floors.

$$K_2 = 0.455 - 0.35\frac{u^{0.75}-1.7}{u^{0.75}+1.7}, \; u = 100\frac{s}{S_r} \tag{2–5}$$

In (2–5) s is the area supported by the element: for plating it is the spacing between stiffeners multiplied by their span, without taking for the span more than *3* times the spacing. By (2–5) the factor K_2 exceeds 0.5 only if $\frac{s}{S_r} < \frac{1}{70}$ and this rarely happens for typical spacing between stiffeners. Hence, for plating, the factor K_2 typically equals *0.5*.

The factor K_3 accounts for the variation of deadrise along the hull, and is given by (2–6), where β is the deadrise angle in degrees at the calculated section.

$$K_3 = \frac{70 - \beta}{70 - \beta_{CG}} \tag{2–6}$$

Table 2-2 summarizes the parameters in our example.

The slamming pressure obtained at the mid-ship of our boat is $p_{sl} = 112\ kPa$.

x/L	K_1	K_2	K_3	s	S_r	u
0.5	*1.0*	*0.5*	*1.0*	*0.106* m^2	*4.67* m^2	*2.26*

Table 2-2 – Parameters related to the design pressure by rules

The slamming pressure is calculated at every $\frac{x}{L}$ between transverse frames (for the calculation of the maximum bending stress at the bottom plating and at the longitudinal stiffeners), as well as at every $\frac{x}{L}$ of a transverse frame (for the calculation of maximum bending stress at the transverse frame). According to the rules, the pressure p_{sl} is applied as a uniformly distributed load to a beam, which represents the local structural member. The end conditions are typically clamped-clamped, as a continuous situation is assumed, where the adjacent spans are applied to similar pressure. To comply with the allowable bending stress *138MPa*, specified by [4] for the boat material *Al5083*, we obtained a required thickness of bottom plate *4.8mm* for a span between stiffeners of *210mm* (at the prismatic boat section from transom to about mid-ship). Toward bow the slamming pressure is increased by pitch (K_1), however decreased by the deeper V (deadrise), and the thickness is sufficient. Figure 2-1 presents the structure of our research boat.

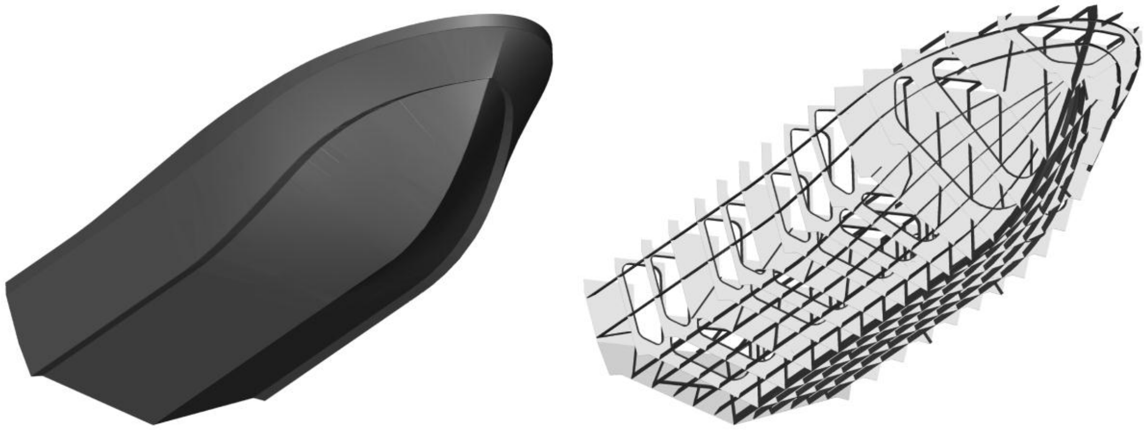

Figure 2-1: Hull Structure, designed by rules

2.2 Theoretical assessment of quasi static pressure

Wagner [7] presented a theoretical solution of the slamming pressure, applied to rigid prismatic wedge entering incompressible water at a constant vertical velocity:

$$p - p_a = \rho V_z \frac{c}{(c^2 - y^2)^{1/2}} \frac{dc}{dt} + \rho \frac{dV_z}{dt} (c^2 - y^2)^{1/2}, c(t) = \frac{\pi V_z t}{2 \tan \beta}, \qquad (2–7)$$

where: ρ is the water density; V_z is the vertical drop velocity, y is a coordinate from the keel across the bottom, c is the wetted half beam, β is the deadrise angle and t is the time from the wetting of the keel. The assumptions of rigid hull and incompressible fluid, result with infinite pressure for a hull of zero deadrise and at $y = \pm c$ for any deadrise. Faltinsen [5] suggested assessing the design pressure by a space-average along the loaded interval, from y_i to y_{i+1} (see Figure 2.2) of Wagner's solution (2–7), assuming a constant vertical velocity. The maximum space-averaged pressure, p_{av}^{max}, occurs when $c = y_{i+1}$. The integration gives a finite average slamming pressure:

$$p_{av}^{max} - p_a = 0.5\rho V_z^2 \frac{\pi}{\tan\beta}\left(\frac{y_{i+1}}{y_{i+1} - y_i}\right)\left(\frac{\pi}{2} - \sin^{-1}\left(\frac{y_i}{y_i + 1}\right)\right) \qquad (2–8)$$

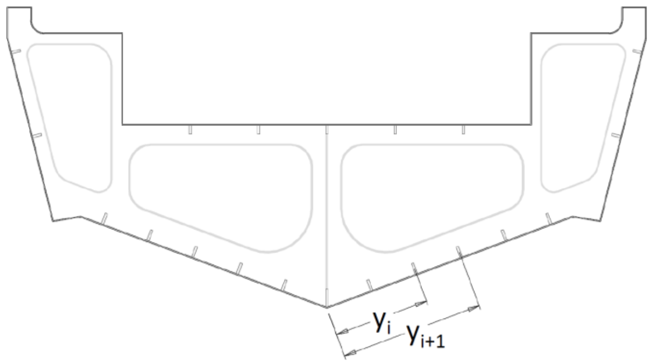

Figure 2-2: Definition of span for space-averaged pressure

2.3 Finding a corresponding drop velocity

The analytical pressure (2–8), as well as our hydro-elastic numerical simulations (presented in the following Item 2.4) are for a vertical drop of prismatic sections into the water (two dimensional problems). The design calculations by classification rules relate the slamming load at each location along the boat to the design service conditions: design speed at a design sea-state, defined by the significant wave height.

Therefore, interpretation of theoretical or numerical results of water entry to the design of a boat requires correlation between the drop velocity and the design service conditions. We define that the vertical velocity of water entry (drop velocity), which corresponds to the design service conditions (speed and sea-state), is the velocity that causes the same slamming pressure: by the theoretical solution (2–8) for a rigid prismatic wedge entering the water at the corresponding drop velocity and by the rules for a boat sailing at the design service conditions (2–3). Both assessments, the theoretical and by the rules, are for a rigid hull (which means quasi static pressure) and are two dimensional (for a finite strip of the boat at specific *x*). Equating (2–3) and (2–8) we obtain:

$$V_z = \left[\frac{140\frac{\Delta}{S_r}K_1K_2K_3a_{CG}}{\rho\frac{\pi}{\tan\beta}\left(\frac{y_{i+1}}{y_{i+1} - y_i}\right)\left(\frac{\pi}{2} - \sin^{-1}\left(\frac{y_i}{y_i + 1}\right)\right)}\right]^{1/2} \qquad (2–9)$$

In our example, the corresponding drop velocity at the mid-ship, is $V_z = 4.0\ m/s$. As the slamming pressure depends on the location along the hull and on the deadrise angle (β), which is increased toward bow, several drop velocities are needed to be found, along the hull.

The concept of the corresponding drop velocity practically accounts for the random nature of the sea. The rules assess a design slamming pressure proportional to the design vertical acceleration, which is related to the significant wave height. According to RINA rules [4], which we adopted, the design vertical acceleration corresponds to the average of the highest *1%* of the random vertical accelerations. By the Rayleigh probability density function (PDF), which is universally used to represent the statistical distribution of wave heights, the average of the highest *1%* of the waves is *1.67* times the significant wave height.

2.4 Hydro-elastic simulations and their implementation to boats design

At Stage 2.3 we obtained for each transvers cross section along the boat, a corresponding drop velocity. A prismatic (two dimensional) rigid section of the hull, with a deadrise as the local deadrise of the boat, vertically entering the water at the corresponding drop velocity, will be applied to the same design pressure calculated by the adopted rules at the design sailing conditions. While the rules calculations apply linear beam theory and are restricted to a rigid hull, as they specify quasi-static pressure; the numerical simulations account for hydro-elastic interactions, dynamics, and nonlinear structural analysis. Thus we need to solve 2D problems of fluid structure interaction, where each section of the boat enters the water at the corresponding drop velocity. As a very fine time stepping is required for the solution of the water-structure interaction, there are no extra efforts in specifying non-linear material and non-linear geometry as well.

While applying the corresponding drop velocity to a deformable hull, we assume that the deformation, which may be important for the local FSI, does not affect the motion of the boat at sea; hence will not affect the correlation between the design service conditions (boat speed and wave height) and the vertical drop velocity, which was obtained for a rigid hull.

In the present study we apply the commercial code ABAQUS/CAE with Arbitrary Lagrangian–Eulerian (ALE) formulation for the fluid domain and Lagrangian formulation for the structure domain. The formulation of the conservation equations of momentum and mass for solving the water domain are presented for example by [8].

2.4.1 The model

The model (Figure 2-3) represents a strip of the bottom plate, under the assumptions:

- The problem is two dimensional (x independent);
- No air entrapment in the water or air cushioning between the water and the boat;
- The problem is symmetrical about the y-z plane, so only half section is modelled.

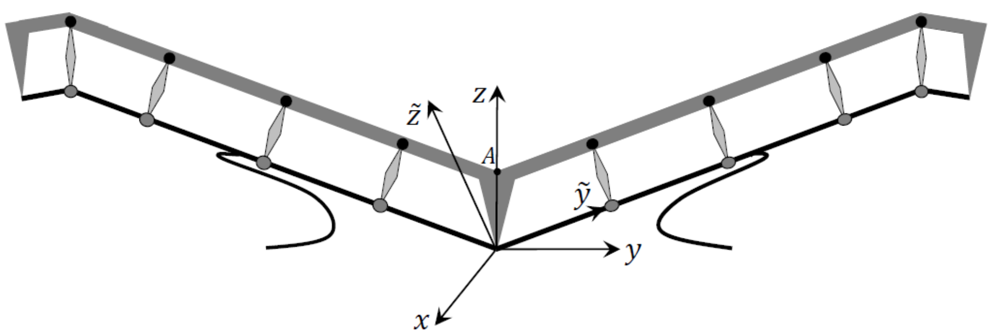

Figure 2-3: Model representation of the boat section

The ALE formulation in ABAQUS does not include a 2D option. The x independency is obtained by using a single element along the boat, and applying Symmetry boundary conditions at the transverse (y-z) planes, aft and forward of the modeled strip.

The boat section is represented by an assembly of three parts: elastic bottom plates; rigid rods, which represent the supports of the bottom plate by the longitudinal stiffeners; and a rigid frame from above (see figure 2-3). The elastic bottom is modeled by four nodes shell elements (*S4R* in ABAQUS). The rigid parts, stiffeners and frame are defined as "rigid bodies". The structure domain includes *1400* elements of size *2mm*. The constraints between the elastic bottom part and the rigid frame are fixed at the keel and at the chine. The inner stiffeners are pinned to the elastic bottom and to the rigid frame.

The fluid domain is modeled by linear Eulerian brick eight nodes elements (*EC3D8R*). The dimensions of the fluid domain are: width *1.6m*, depth *0.4m* and thickness (along x) *2mm* (element size). An additional grid of height *0.2m* is included above the water line for the run-up. For the element size of $2\times2\times2mm$ we obtained *240,000* elements in the water domain.

2.4.2 Boat Material

An experimental strain-stress curve, transformed to "true" strain, presented by Figure 2-4, was used as the material constitutive law. Table 2-2 presents the boat material properties.

Material Aluminum 5083 *H321*	
Young's modulus *E*	52.9*MPa*
Poisson ratio *ν*	*0.3*
Density *ρ*	$2700kg.m^{-3}$

Table 2-2: Boat material properties

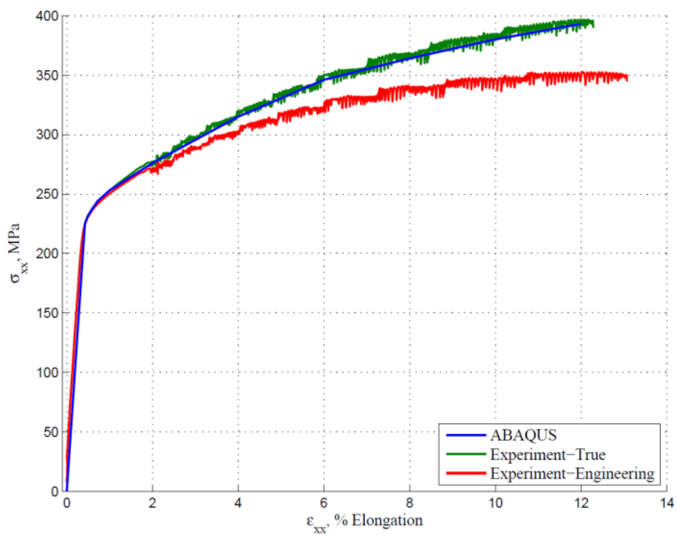

Figure 2-4: Engineering and True stress strain curve

2.4.3 Database for design

This section presents results of the simulations of water entry, over a wide range of parameters, which we consider sufficient for a practical range of design. The results map the stresses over transverse sections of the bottom plate, for different deadrise angles, plate thicknesses, spans between stiffeners and velocities of water entry. We carried out a systematic series of 225 simulations by programing a parametric script in Python (the script language used by ABAQUS). Table 2-3 presents the varying parameters.

Figure 2-5 presents an example of the pressure distribution along the structure-water interface, together with the structure deformation at two times during the water entry.

Variable	***units***	***Cases***
Drop velocity – V_z	$\mathrm{m.s^{-1}}$	*(-3, -4, -5)*
Length between stiffeners – L	mm	*(210, 315, 420)*
Bottom Plate thickness – d	mm	*(3.2, 4.0, 4.8, 6.0, 8.0)*
Dead rise angle – β	degree	*(7, 14, 21, 28, 35)*

Table 2-3: Varying parameters of the simulated cases

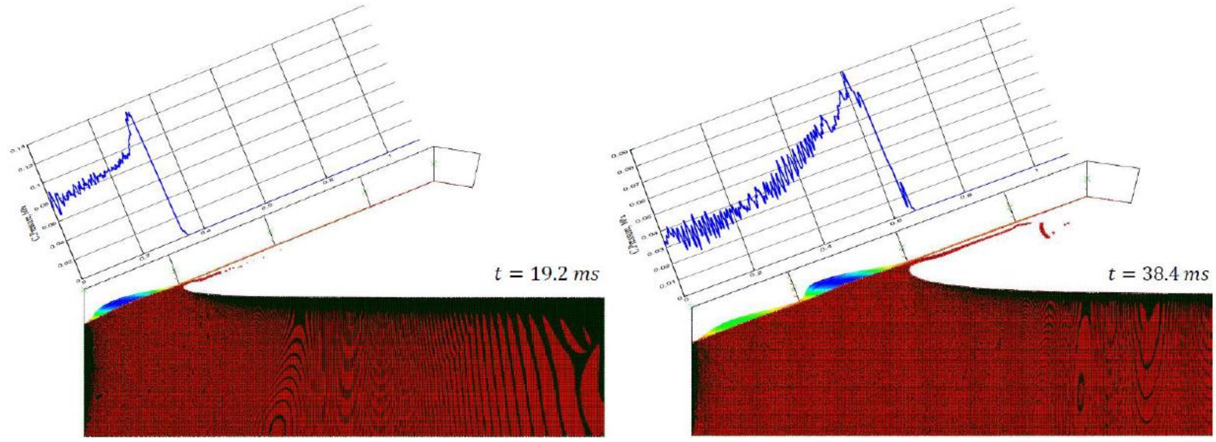

Figure 2-5: Water surface, structure deformation and distribution of contact pressure at successive times of water entry, for a drop velocity $V_z = -4\ m.s^{-1}$, $\beta = 21^o$ and $L = 315\ mm$

At the first time presented, the jet location is at the first stiffener. As the second span is not loaded, the right boundary condition for the first span is more like a simple support and not a clamp, as assumed by the rules. At a later time, loading of the second span make the first span to deform similar to a clamped-clamped beam. Modeling the bottom plate as a continuous beam supported by the stiffeners is more representative than the rules calculations and does not require an assumption regarding the end conditions for each span.

As the pressure distribution varies dramatically during the water entry, it is not trivial to guess the time step at which the load effects are most critical. Hence, for each analysis a maximum strain envelope (maximum over time for each location) is stored and presented. As the simulation time is long and 225 cases were processed, we limited the duration of the solution to the water entry of the middle of the second span. Simulations for the whole process show that the maximum strain envelope at the first span is not varied while continuing the simulation. Figure 2-6 presents examples of the strain envelopes for the whole process of water entry.

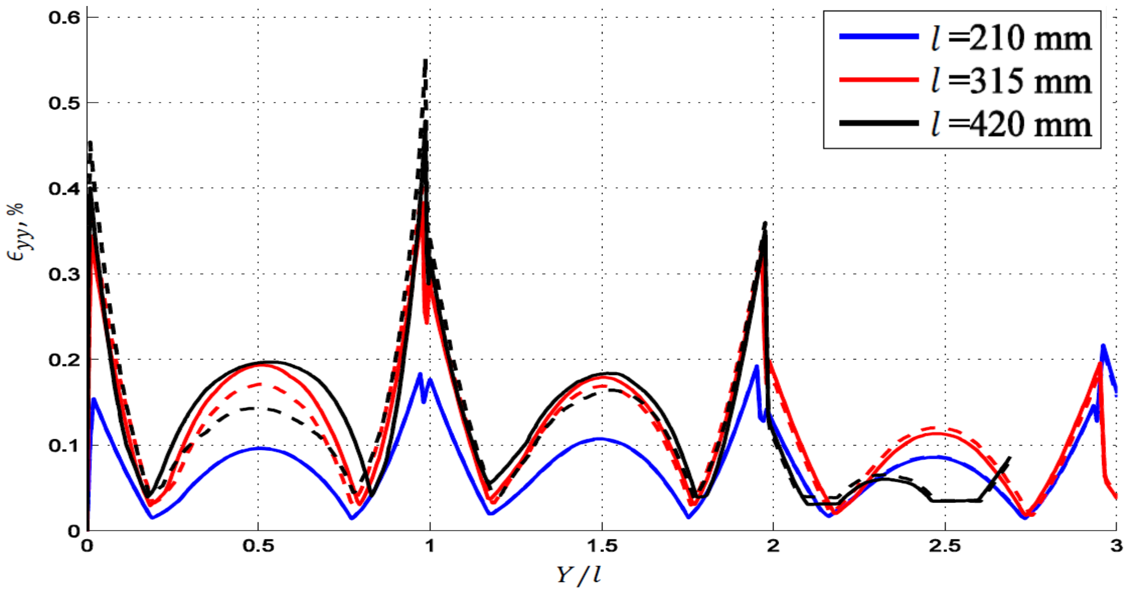

Figure 2-6: Maximum strains envelop (over time) along three spans, for $V_z = -4\,m.s^{-1}$, $\beta = 21^o$ and $d = 4\,mm$. Strains at the top and the bottom of the plate are shown by solid and dashed lines respectively

Storing the maximum stresses over the time and location allow us to present a database of 225 simulations in a compact manner, usable for design. The results are gathered in three figures in [10], one for each velocity of water entry. Each figure presents 75 cases: *5 deadrise angles* × *5 thicknesses* × *3 spans*. Angles of deadrise are distinguished by colors, spans by symbols and plate thicknesses by the horizontal axis. Calculation by linear beam theory, assuming theoretical uniformly distributed load and clamped-clamped end conditions (as typically assumed by boat design rules) are shown in continuous lines of different line-styles for different spans. These lines are trimmed at the yield stress, above which the linear theory is not valid. A verification of the simulations is presented by approaching the static-linear theory when the structure becomes rigid (thick plates and short spans).

As an example we may apply the hydro-elastic design charts to optimize the boat, for which we presented the design by rules. We use the design chart for the corresponding drop velocity that we obtained at mid ship, $V_z = 4.0$ *m/s*, which is presented by Figure 2-7. A blue line presents the allowable stress of *138MPa*. It is shown on the figure that for a deadrise angle of *21°* the rules method results with a required plate thickness of *4.2mm*, while only *3.4mm* is required by the rational approach; a reduction of *20%*. The research side of our boat has plate thickness of *4mm* and the spacing between stiffeners is *420mm*. By Figure 2-7 we obtain a maximum stress of *230MPa*. For such high stresses, we will obtain more significant differences between rules calculations and rational analysis, as we present in section 4.

3 DESIGN AND CONSTRUCTION OF THE RESEARCH BOAT

We designed and constructed a full scale research boat and held sea trials. Our research boat, named Dganit, has two sides of different construction: the port side is designed by rules with bottom plates of *4.8mm* stiffened longitudinally at spaces *210mm*, while the starboard side is designed by our rational design method, with *4.0mm* plates and double spacing, *420mm*, between the stiffeners. Figure 3-1 presents the construction at B.M. Carmel.

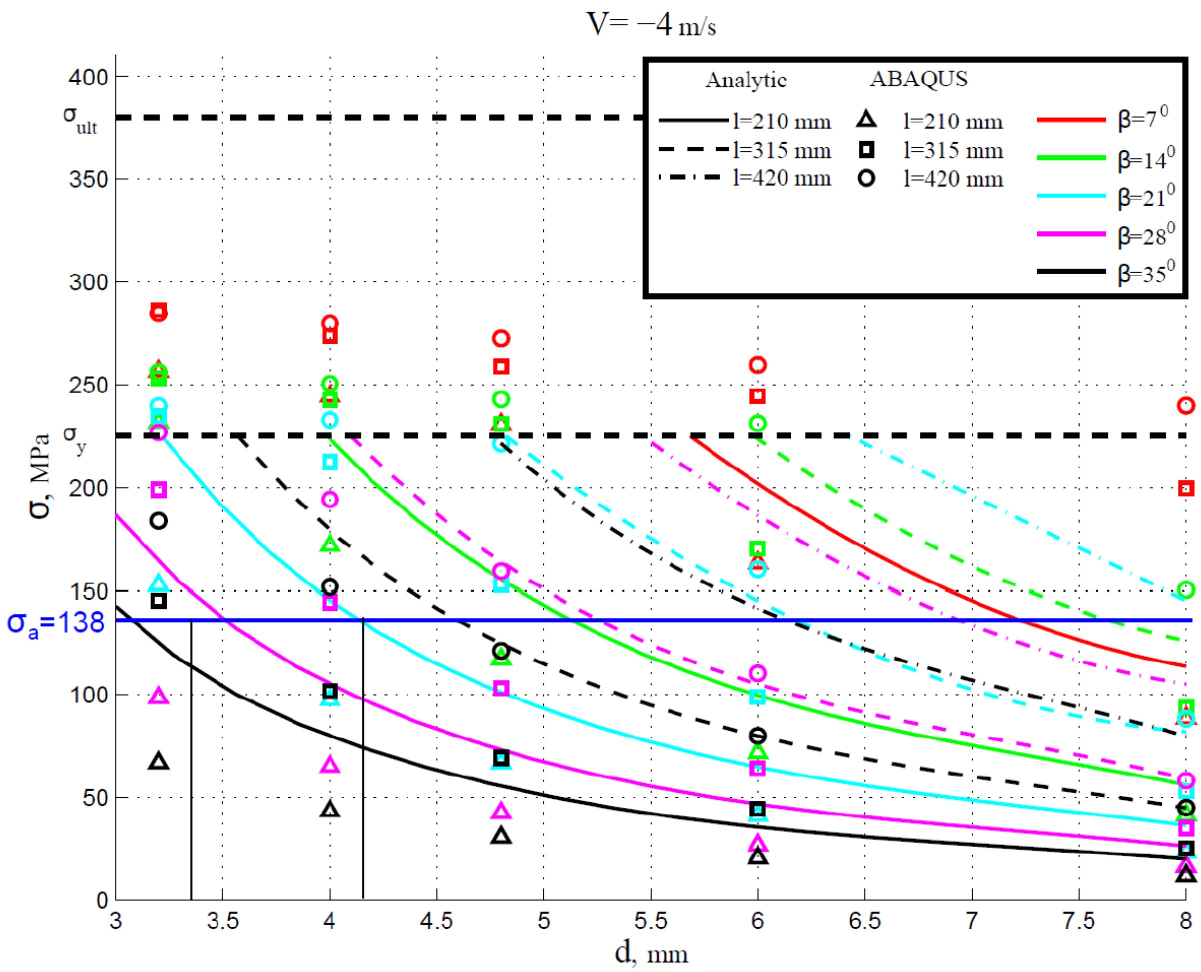

Figure 2-7: Flow chart for the design example

Figure 3-1: Research boat construction at B.M. Carmel

The boat is equipped with an array of strain gages, located at points of high stresses (see Figure 4-3). The Stored Sampling rate is *2048Hz*, which is processed by filtration of raw sampling at *40,000Hz*.

4 PRELIMINARY RESULTS OF THE SEA TRIALS

Figure 4-1 presents our research boat at sea trials. Figure 4-2 presents typical strain records. As the sea is random, the comparison requires statistical processing of the measurements. Adopting the rules approach, we process the strain records of each leg, during which the sea state is assumed stationary, to find the *mean of the highest 1%* of stresses.

Figure 4-1: Research boat Dganit during sea trials

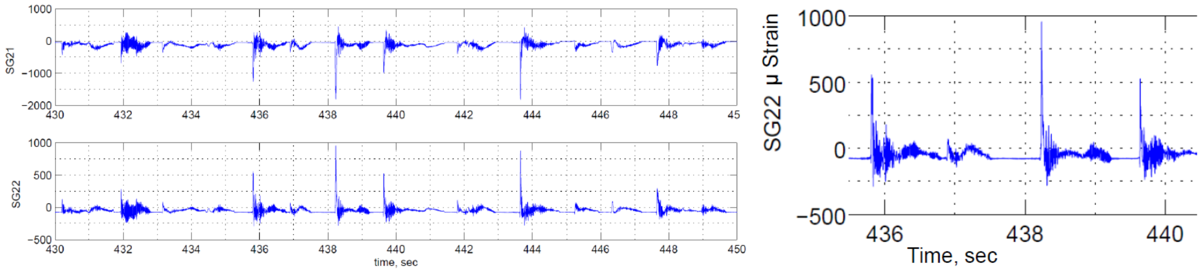

Figure 4-2: Example of Strains records at *26knots* Head seas *Hs 1.1m*

Figure 4-3 presents comparison of stresses between design by rules, our rational design, and trials measurements. The results are presented at spots of highest stresses, at the ends of spans. Due to the fillet welds, the strain gages are locates about *20mm* from these spots and the measurements are factored to represent the maximum stresses. The factors are obtained from the strain envelopes by analysis. The comparison clearly shows: For the heavy side (designed by rules), rules, rational, and trials show similar stresses; however, for the light side, the rules dramatically over assess the stresses, while rational and trials are similar.

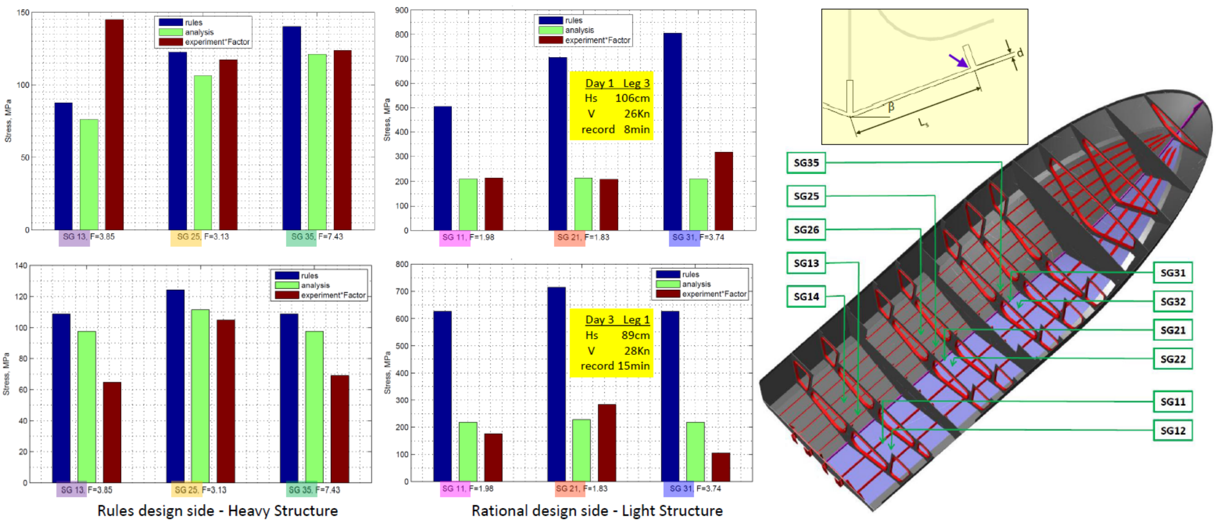

Figure 4-3: Comparison of stresses

5 CONCLUSIONS

This paper presents a rational design method for the structure of planing hulls, which considers hydro-elasticity and dynamic nonlinear structural response. A database of results of simulations of water entry, in a wide range of parameters, enable assessment by designers, without the need to setup and run FSI simulations.

Preliminary processing of measurements of sea trials of a full scale research boat, validate the method and show that for a light structure it is significantly more pragmatic than design by rules.

Currently, the method is extended to include a Fatigue Limit State (FLS) design procedure. We expect that this study will advance the design and production of more efficient boats.

ACKNOWLEDGEMENTS

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